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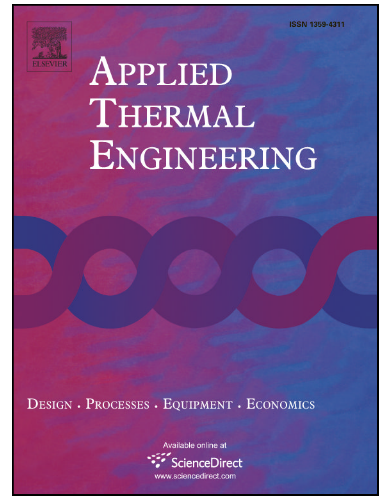

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Comparisons between Heat Pipe, Thermoelectric System, and Vapour Compression Refrigeration System for Electronics Cooling

Kun Liang*, Zhaohua Li, Ming Chen, Hanying Jiang

Department of Engineering and Design, University of Sussex, Brighton BN1 9QT, UK

* Corresponding author. Tel: +44 1273 678573. Email address: kun.liang@sussex.ac.uk

Abstract

Passive systems such as air for electronics cooling have now effectively reached their limits. This paper evaluated three comparable systems for electronics cooling, including heat pipe (HP, passive system), thermoelectric (TE) and vapour compression refrigeration (VCR) systems (active systems). Mathematical model has been built for the heat pipe and the thermoelectric system respectively. Measurements have been conducted to validate the model and to compare the performance among a HP, a single stage TE system and a two-stage TE system, a combination of the HP and the TE system, and a VCR system using an oil-free linear compressor. Close agreements between the modelling and measurements have been achieved in terms of electric power input and cooling capacity at various temperatures. The HP improved the cooling capacity and the coefficient of performance (COP) of the TE system by 53% and 42% respectively at a cold end temperature of 10 °C. Heat pipe is more attractive for cooling large devices at higher temperatures. Two-stage TE system can be used for cooling devices at lower temperatures. VCR system is capable of dissipating much higher heat flux (200 W/cm^2) at lower temperature than all other technologies.

Keywords: thermoelectric, heat pipe, vapour compression refrigeration, two-stage, COP, cooling capacity

NOMENCLATURE

Symbols

A	area (mm^2)
COP	coefficient of performance*d* sphere diameter (mm)
d	outer diameter (mm)
f	thermoelectric module packing fraction
h	convective heat transfer coefficient (W/m^2K)
I	input current (A)*K* thermal conductance (W/m^2K) or wick permeability (mm^2)
k	thermal conductivity (W/mK)
l	length of thermoelement or heat pipe (mm) or latent heat (kJ/kg)
N	number of thermocouples
P	power input (W)
$\dot{Q}$	cooling capacity (W)
R	electrical resistance (Ω) or thermal resistance (Km^2/W)
RMS	root mean square
r	radius of wick (mm)
T	temperature (°C)

Acronyms

AC	alternating current
CCD	charge-coupled device
CPU	central processing unit
DAQ	data acquisition
DC	direct current
HP	heat pipe
HVAC	heating, ventilation, and air conditioning
IR	infrared
PM	power meter
PR	pressure ratio
PT	pressure transducer
PTC	positive temperature coefficient
TC	thermocouple
TE	thermoelectric
V	voltage
VCR	vapour compression refrigeration

Greek letters

α	Seebeck coefficient
ρ	electrical resistivity or density
φ	angle of heat pipe relative to the vertical axis
σ	surface tension
μ	viscosity
ε	porosity

Subscripts

ad	adiabatic
b	boiling

c	cooling or cold end or condenser
e	element or evaporator
eff	effective
f	fluid
h	hot end
i	inner
max	maximum
o	outer
t	total
w	wick

1. Introduction

The power density of today's electronic devices is accelerating as a result of an increasing number of transistors in silicon chips. According to Moore's Law, the number of transistors on integrated circuits doubles approximately every two years. This trend has slowed in recent years but the transistor density is still increasing. The International Technology Roadmap for Semiconductors **[1]** predicts that by 2022 the average power consumption for a typical stationary chip to be over 400 W and over 800 W by 2026. As a result of the concentration of chip components within a small area, together with the power required to operate these electronic circuits, the quantity of heat generated by chips has correspondingly increased. This heat must then be dissipated in order to maintain the chip at an acceptable operating temperature, generally acknowledged to be less than 85°C, although hot spots within the die area may reach much higher temperatures **[2]**. Passive cooling systems such as air, liquid and heat pipe (HP) have now effectively reached their limits. Therefore, if heat generation by chips is to increase further in the future, more effective cooling methods will be needed **[2]**. Agostini et al. **[3]** reviewed different technologies for heat removal of computer chips and concluded that boiling in micro-channels is the technology with the greatest potential.

Active cooling systems such as vapour compression refrigeration (VCR) system can achieve low junction temperatures while dissipating high heat fluxes **[4]**. The miniature VCR system typically consists of miniature compressor and microchannel heat exchangers (evaporator and condenser). Linear compressor can be attractive for this system due to possible oil-free operation and miniaturization. The main concern is the cost. A significant number of studies have been carried out in the past ten years or so regarding miniature and small scale refrigeration systems for electronics cooling. Trutassanawin et al. [**5, 6, and 7**] conducted numerical and experimental investigations of a miniature VCR system using R134a. Sathe et al. [**8**] experimentally evaluated a miniature rotary compressor for application in electronics cooling. Mongia et al. [**9**] developed an R600a (isobutane) prototype VCR system for portable (notebook) computers using a 12V DC reciprocating compressor and micro-channel heat exchangers. A miniature VCR system was built employing a moving magnet linear compressor by Possamai et al. [**10**].

Among active cooling systems, thermoelectric (TE) cooling system is considered to be an alternative technology that has advantages of high reliability, no moving parts, compact in size, light weight and no working fluid **[11]**. A thermoelectric module consists of a bunch of thermocouples wired electrically in series and thermally in parallel. Each thermocouple is made of two different semiconducting thermoelements (N- and P-type), which generate thermoelectric cooling effect (Peltier-Seebeck effect). When a DC voltage is applied through the connected junction, one end is cooled and the other is heated. The hot end must be cooled to avoid damage **[12]**. The main disadvantages of the thermoelectric cooler are the high cost and low energy efficiency. A number of studies have been conducted to improve the performance of the TE system. Riffat and Ma **[13]** concluded that the optimum TE module design will be a compromise between the requirements of coefficient of performance (COP) and heat pumping capacity and for large temperature difference applications, the COP can be improved significantly by use of multistage thermoelectric modules. Lindler **[21]** concluded that at a hot side temperature of 100°C, the single TE module COP is 0.151 compared to 0.232 for the cascaded module which represents a 54% improvement in COP. Chen et al. **[22]** also shows that when the temperature ratio of the heat sink to the cooled space is larger, both the maximum COP and the cooling capacity of a two-stage TE are larger than those of a single-stage system. Huang et al. **[14]** investigated the thermal performance of a thermoelectric water-cooling device for electronic equipment. Liu et al. **[15 and 16]** proposed a TE coupled with a micro heat pipe system for cooling CPU. Results show that an operating voltage of 12 V could achieve the maximum cooling capacity and heat pipe significantly improves the cooling performance of the TE system. Sun et al. **[17]** proposed a thermoelectric cooling system coupled with a gravity-assisted heat pipe for cooling electronic devices.

This work evaluated the performances of five different systems for electronics cooling, including a single stage TE system, a two-stage TE system, a HP, a combination of TE and HP, and a VCR system using an oil-free linear compressor. A mathematical model has been built for the HP and the TE system, respectively. The HP model is based on capillary limitation. Experimental studies have been conducted to validate the models and to quantitatively compare the performance among these cooling technologies.

2. Modelling of the TE System and the HP

2.1 Modelling of the Thermoelectric (TE) System

The one-dimensional model of the TE ignores the radiation and convective heat transfer between the thermocouple and the ambient air. The model will predict the electric power input and cooling capacity at different operating conditions (input current and cold end temperature).

The power input *P* can be calculated as follow

$$P = I^2 R + \alpha I (T_\mathrm{h} - T_\mathrm{c}) \tag{1}$$

where I is the input current, R is the electrical resistance, α is the thermoelectric module Seebeck coefficient, T_h is the hot end temperature and T_c is the cold end temperature.

The cooling capacity $\dot{Q}_\mathrm{c}$ can be calculated according to

$$\dot{Q}_\mathrm{c} = \alpha I T_\mathrm{c} - K(T_\mathrm{h} - T_\mathrm{c}) - 0.5 R I^2 \tag{2}$$

where *K* is the thermal conductance.

For a given thermoelectric material, following equations developed by Chen and Snyder **[18]** can be used to calculate the thermoelement Seebeck coefficient α_e, electrical resistivity ρ and thermal conductivity k

$$\alpha_\mathrm{e} = \frac{\dot{Q}_\mathrm{c,max}(T_\mathrm{h} - \Delta T_\mathrm{max})}{N {T_\mathrm{h}}^2 I_\mathrm{max}} \tag{3}$$

$$\rho = \frac{Af(T_\mathrm{h} - \Delta T_\mathrm{max})^2}{2{T_\mathrm{h}}^2 l} \frac{\dot{Q}_\mathrm{c,max}}{N^2 {I_\mathrm{max}}^2} \tag{4}$$

$$k = \frac{l(T_{\mathrm{h}} - \Delta T_{\max})^2}{A f {T_{\mathrm{h}}}^2} \frac{\dot{Q}_{\mathrm{c,max}}}{\Delta T_{\max}} \tag{5}$$

where N is the number of thermocouples, A is the cross-sectional area, f is the thermoelectric module packing fraction, l is the length of thermoelement, and $\Delta T_{\max}$ is the maximum temperature difference between the cold and hot ends.

Therefore, α, R and K for the thermoelectric module can be calculated as follow

$$\alpha = N\alpha_{\mathrm{e}} \tag{6}$$

$$R = \frac{N^2 l \rho}{A f} \tag{7}$$

$$K = k\frac{A f}{l} \tag{8}$$

The hot end temperature T_{h} will be calculated from the modelling of the HP.

The COP can be calculated as

$$\mathrm{COP} = \frac{\dot{Q}_{\mathrm{c}}}{P} \tag{9}$$

2.2 Modelling of the Heat Pipe (HP)

The heat pipe is a vapour–liquid phase-change device that transfers heat from a hot reservoir to a cold reservoir using capillary forces generated by a wick or porous material and a working fluid. Due to the two-phase characteristics, heat pipe is ideal for transferring heat over long distances with a very small temperature drop and for creating a nearly isothermal surface for temperature stabilization **[19]**. This is very attractive for reducing the hot end of the TE.

For most heat pipes, the cooling capacity $\dot{Q}_{\mathrm{c}}$ due to capillary limitation can be expressed as **[19]**

$$\dot{Q}_{\mathrm{c}} = \frac{\sigma \rho l_{\mathrm{f}}}{\mu} \frac{K A_{\mathrm{w}}}{l_{\mathrm{eff}}} \left(\frac{2}{r_{\mathrm{eff}}} - \frac{\rho g l_{\mathrm{t}} \cos\varphi}{\sigma}\right) \tag{10}$$

where σ is the surface tension, ρ is the density of the working fluid, μ is the viscosity, l_{f} is the latent heat of the fluid, K is the wick permeability, A_{w} is the cross-sectional area of the wick, l_{t} is the total length of the heat pipe, r_{eff} is the effective radius of wick, and l_{eff} is the effective length of the heat pipe.

The wick permeability can be calculated as follow

$$K = \frac{d^2 \varepsilon^3}{150(1-\varepsilon)^2} \tag{11}$$

where d and ε are sphere diameter and porosity respectively.

The effective length of the heat pipe is

$$l_{\text{eff}} = 0.5(l_{\text{e}} + l_{\text{c}}) + l_{\text{ad}} \tag{12}$$

where l_{e}, l_{c} and l_{ad} are length for evaporation, condensation and adiabatic sections respectively.

2.3 Combination of the Heat Pipe and Thermoelectric Cooler

When the HP and the TE system are combined, the cooling capacity of the HP at the evaporation zone equals to the heat released from the hot end of the TE. Therefore, the hot end temperature of the TE T_{h} can be calculated according to

$$\dot{Q}_{\text{c}} = \frac{l_{\text{e}}}{R}(T_{\text{h}} - T_{\text{b}}) \tag{13}$$

where T_{b} is the boiling temperature of the working fluid and R is the thermal resistance of the evaporation zone, which can be defined as below

$$R = \frac{1}{h\pi d_i} + \frac{1}{2\pi k}\ln\left(\frac{d_o}{d_i}\right) \tag{14}$$

where h is the convective heat transfer coefficient, k is the thermal conductivity of the pipe material, d_o is the outer diameter of the evaporation zone and d_i is the inner diameter.

The combination of the HP and TE system is shown in Fig. 1. The hot end of the TE was attached to the evaporation zone of the HP by using thermal grease. The condensation zone of the HP was connected to a series of fins as a heat sink. An axial fan above the heat sink can dissipate the heat to the ambient. The cold end of the TE can be directly attached to the CPU using thermal grease. When a DC voltage is applied to the TE, there is a temperature difference between the hot and cold ends due to the electron flow. The heat from the hot end of the TE is transported by the working fluid in the HP from the evaporation zone to the condensation zone. This will significantly reduce the temperature difference between the hot and cold ends of the TE.

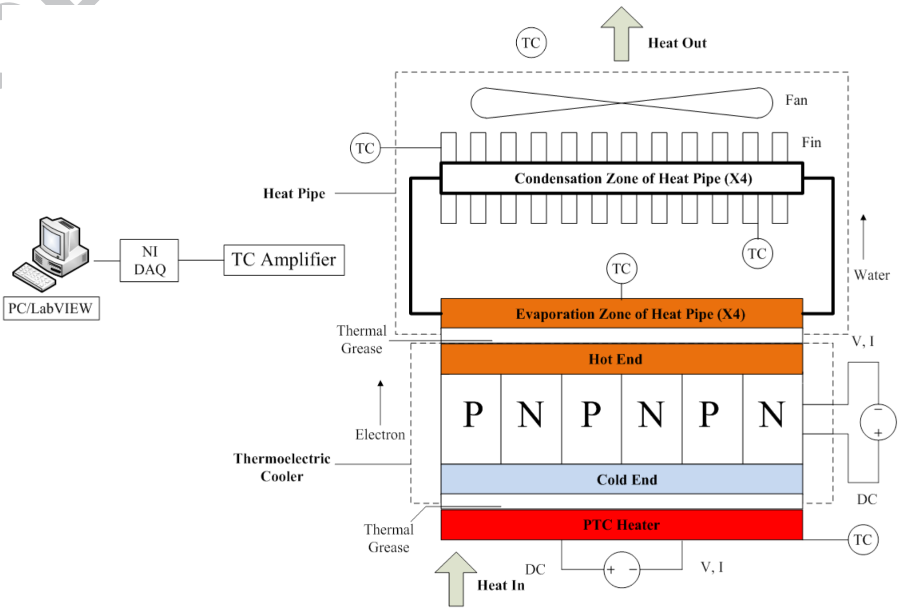

Fig. 1 Schematic of the combination of the heat pipe and thermoelectric cooler, and the instrumentation. TC: thermocouple; V: voltage; I: current; DAQ: data acquisition

3. Measurements of the VCR System, HP, and TE System

3.1 Vapour Refrigeration System

Among the available alternatives, VCR is one of the only technologies which can work in high-temperature ambient, and even result in negative values of thermal resistance. The advantages of refrigeration cooling include maintenance of low junction temperatures while dissipating high heat fluxes, potential increases in microprocessor performance at lower operating temperatures, and increased chip reliability. The proposed VCR system for electronics cooling is composed of an oil-free linear compressor, a compact condenser, a throttling device, and a compact evaporator, as shown in Fig. 2. The refrigerant for the system is R134a. The details of the oil-free linear compressor have been reported in [**20**].

Performances of the proposed VCR system were measured using an instrumentation and data acquisition system shown in Fig. 2. The heat into the evaporator was controlled manually by a variac to simulate the CPU heat source. Three Druck PMP1400 pressure transducers were used to measure the compressor discharge pressure, evaporator inlet pressure, compressor suction pressure. Five K-type thermocouples were employed to measure temperatures including the: compressor discharge, condenser outlet, evaporator inlet, compressor suction, and evaporator outlet. The mass flow rate was measured by a mass flow meter (Hastings HFM-201). The mass flow meter is thermal-type which uses a bypass (proportional to the total flow) and the thermal properties of a gas to infer the mass flow rate, with an output voltage from 0 to 5 V. The power input to compressor, overall RMS current and voltage can be read from a power analyser PM100. An AC power amplifier (Vonyx VXA-2000) amplified the analogue signal from the data acquisition system in order to drive the linear compressor. Two data acquisition cards USB 6351 from National Instruments were used for the measurements. The waveform signal to drive the compressor was generated from the analogue output. A LabVIEW programme has been written to display and log the temperatures during the measurements.

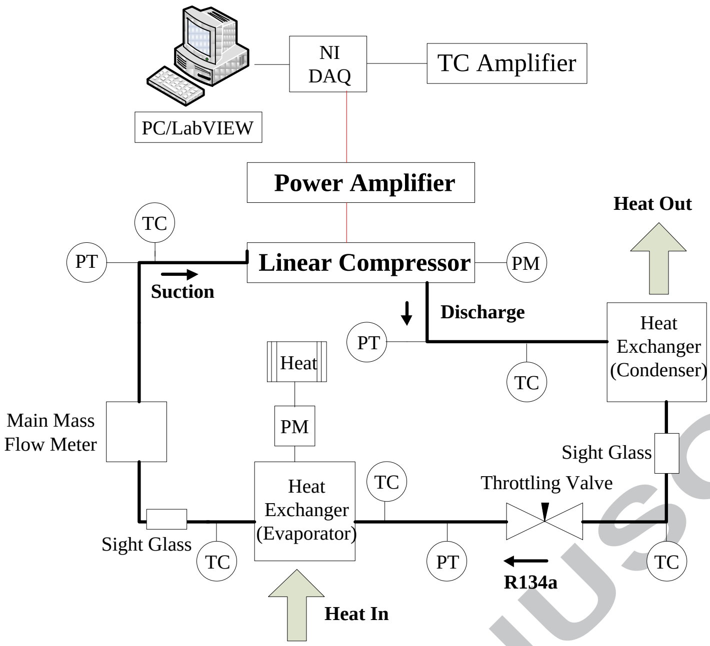

Fig. 2 Schematic of the vapour compression refrigeration system and the instrumentation. PT: pressure transducer; PM: power meter; TC: thermocouple; DAQ: data acquisition

Measurements for the VCR system have been conducted at different pressure ratios and compressor strokes at a condenser outlet temperature of 50 °C.

3.2 Heat Pipe and Thermoelectric System

The off-the-shelf single-stage thermoelectric module was TEC1-12706 with dimension of 40 mm x 40 mm x 3.75 mm and material of bismuth tin (Bi-Sn). Maximum temperature difference $\Delta T_{\max}$ is 67 K and maximum cooling capacity $\dot{Q}_{\mathrm{c,max}}$ is 60 W. A radiator with dimension of 116 mm x 100 mm x 23 mm was used when only TE was used for cooling. Four direct contact copper/water heat pipes combined with 0.5 mm thickness aluminium fins have flat surface for evaporation zone allowing minimum thermal contact resistance. The outer diameter is 6 mm. The length for evaporation, condensation and section against gravity are 35 mm, 240 mm and 70 mm respectively. A 12 V DC fan (flow rate of 0.03 m^3/s) was used to dissipate the heat from the heat sink. The 2-stage TE is also an off-the-shelf TE-2-(127-127)-1.15 with a maximum current of 6 A and a $\Delta T_{\max}$ of 100 K. Multi-Stage modules are modules that are stacked or cascaded to achieve higher temperature differences than cannot be achieved by single-stage modules.

Performances of the HP, TE and their combination were measured using an instrumentation and data acquisition system shown in Fig. 1. An enclosure PTC (positive temperature coefficient) heating element was used to simulate the CPU heat source. The maximum power output is 80 W and the dimension is 75 mm x 35 mm x 8.5 mm. Thermal grease was filled between the PTC heater and the cold end of the TE. The heat source was insulated by foam plastic to avoid radiation. The heat/cooling capacity of the system was monitored and controlled by the DC power supply. Another DC power supply was used for the TE to monitor and control the voltage applied to the TEC. Four K-type thermocouples (accuracy of 2 K) were used to measure the temperatures of cold end, hot end, heat sink and ambient. As the original thermocouples only produce a few mV output, an AD595 amplifier was used. The thermocouples were calibrated by comparing their output voltages to known reference temperatures provided by mercury in glass thermometer with 0.2 K resolution. A data acquisition card USB 6351 from National Instruments was used for the measurement. A LabVIEW programme has been written to display and log the temperatures during the measurements with sampling rate of 200 Hz.

Measurements have been conducted for four systems: single stage TE with radiation and fan, HP with fin and fan, combination of HP and TE, and 2-stage TE with radiation and fan. The voltage supplied to the TE ranged from 3 V to 15 V. Cold end temperature of the TE ranged from -10 °C to 90 °C. Ambient temperature was 19 °C. The experimental results were used to validate the model and to compare the performance of all five systems.

4. Results and Discussions

4.1 Model Validation

The cooling capacity for the heat pipe against cold end temperature is plotted for both modelling and measurements in Fig. 3. Note that for this test the evaporation zone of the heat pipe was directly attached to the PTC heater so that the cold end temperature means the evaporation zone temperature. Close agreement can be seen with average error of 8%. At a cold end temperature of 55 °C, the cooling capacity is 100 W and the fin temperature is 32 °C. The heat flux that can be dissipated for a 2 cm^2 chip at 80 °C is 65 W/ cm^2.

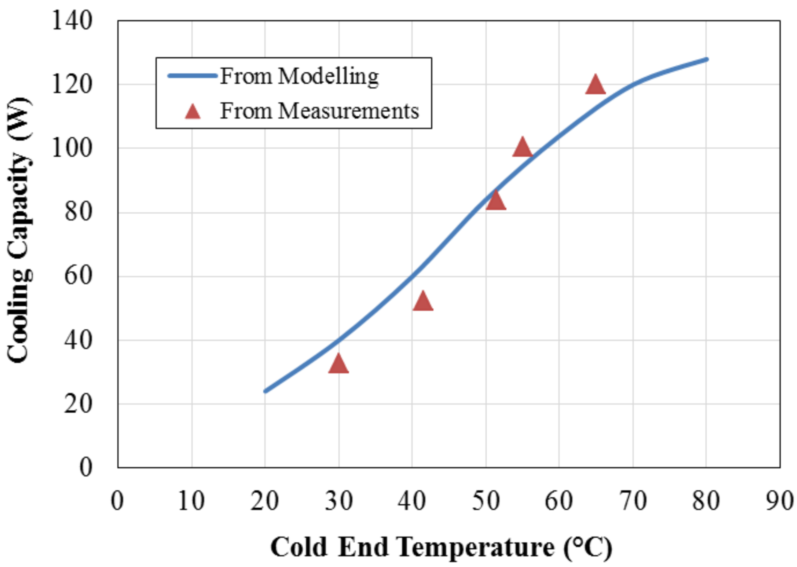

Fig. 3 Performance of the heat pipe from modelling and measurements

Fig. 4 compares the power input and the COP between modelling and measurements at a voltage of 15 V for the single stage TE system. Power input from modelling agrees well with measurements across the range of the cold end temperature. The error percentage is only 3%. The discrepancy for the COP averages at 18%.

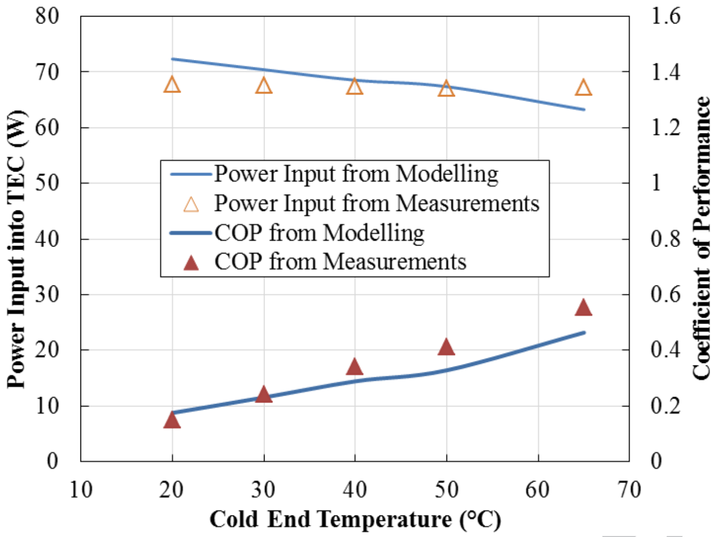

Fig. 4 Comparison of power input into the TE and COP between modelling and measurements with a voltage of 15 V

4.2 Performance of the Thermoelectric (TE) System from Measurements

The cooling capacity from measurements against temperature difference between the cold and hot ends of the TE is shown in Fig. 5 at different voltage. Cooling capacity increases linearly with the decrease of the temperature difference as can be expected from Equation (2). The similar slope for each voltage indicates that the thermal conductance *K* appears to be independent of current and temperature. With a fixed temperature difference, generally higher voltage leads to higher cooling capacity and 10 V is the optimal voltage as it provides higher cooling capacity than 12 V.

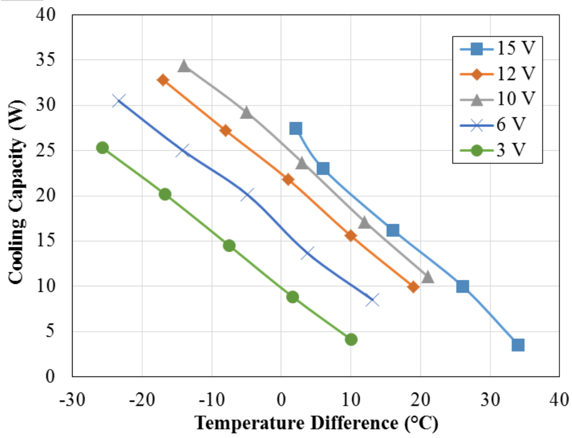

Fig. 5 Cooling capacity against temperature difference between cold and hot ends of the TE at different voltages

Fig. 6 shows that the COP for the TE also increases with the decrease of temperature difference. Lower voltage leads to much higher COP and also much higher rate of increase against decreasing temperature difference. At a temperature difference of 10 °C, the COP is 1.3 for 3V and 0.33 for 12 V. However, 12 V provides 4 times the cooling capacity of 3 V. It can be seen that 10 V gives reasonable COP (0.33-1.24) across the temperature difference range.

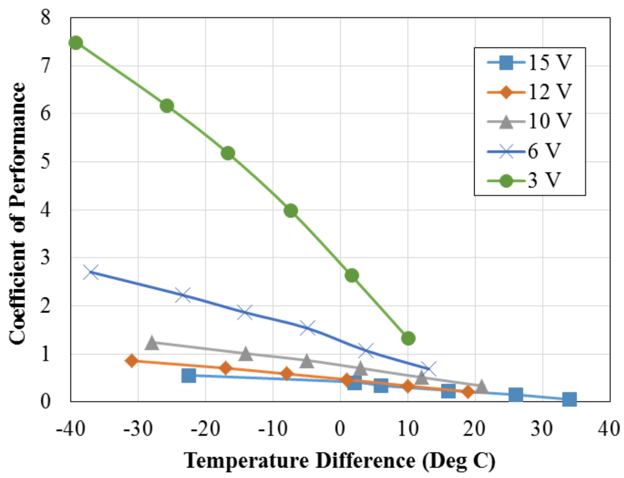

Fig. 6 COP against temperature difference between cold and hot ends of the TE at different voltages

Fig. 7 shows that the cooling capacity linearly increases with the cold end temperature and 10 V provides the highest cooling capacity across the cold end temperature range. The lower cooling capacity at 12 V and 15 V is due to much higher hot end temperature (over 40 °C). At a cold end temperature of 80 °C and hot end temperature of 39 °C, 10 V can provide 49 W cooling for the CPU with a CoP of 1.5. This is typical CPU cooling condition.

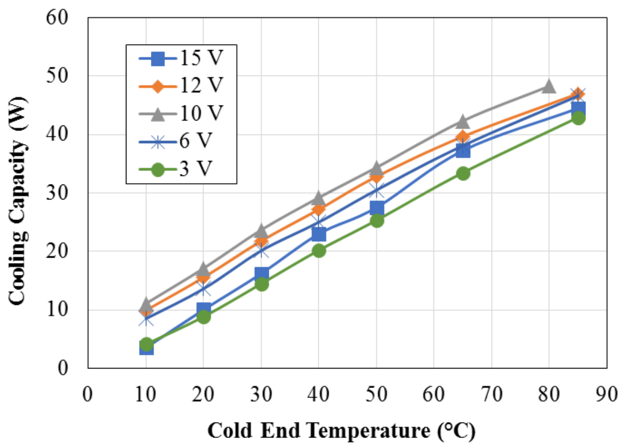

Fig. 7 Cooling capacity against cold end temperature of the TE at different voltages

4.3 Performance of the VCR System from Measurements

Pressure ratio and compressor stroke range from 2.5 to 3.5 and from 10 to 13 mm respectively. The condenser outlet temperature was maintained at 50 °C. The evaporator temperature varies from 6 to 26 °C. Fig. 8 shows the power input into the linear compressor of the VCR system against the mass flow rate of R134a. Power input increases linearly with mass flow rate for each pressure ratio. Pressure ratio of 2.5 corresponds to evaporator temperature of 20 °C. To deliver more mass flow, higher stroke is required for smaller dead volume. For a mass flow of 3.3 g/s, power input required is 150 W at peak stroke (13 mm).

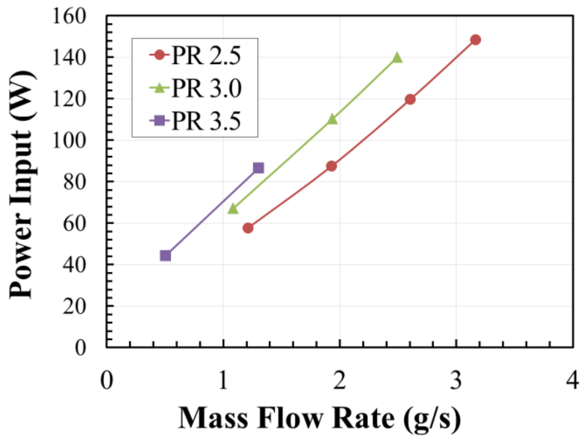

Fig. 8 Power input into the VCR system against mass flow rate of R134a at different pressure ratios

The cooling capacity and the COP of the VCR system against evaporator temperature are shown in Fig. 9. For evaporator temperature of 20 °C, the cooling capacity is 400 W and the COP is 3.2. This is a reasonably good performance. For a CPU with die size of 2 cm^2, the heat flux that can be dissipated is 200 W/ cm^2 at a junction temperature of 20°C. Cooling capacity and COP decrease with decreasing evaporator temperature.

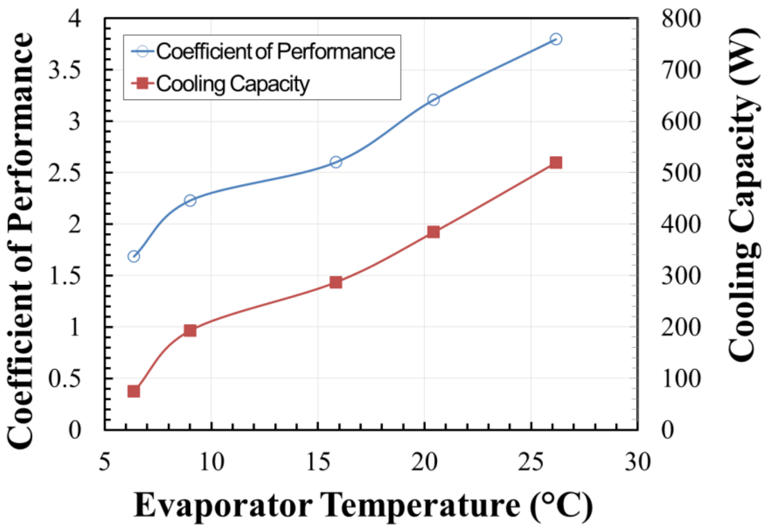

Fig. 9 Cooling capacity and COP of the VCR system against evaporator temperature at a stroke of 12 mm and condenser outlet temperature of 50 °C

4.4 Comparisons between the HP, TE, combined TE and HP, 2-stage TE, and VCR

Fig. 10 compares the performance of the combined HP and TE with TE with a voltage of 12 V. The model of the combined HP and TE predicts very close cooling capacity to the measurements. The cooling capacity at a cold end temperature of 80 °C for TE and combination of HP and TE are 46 W and 58 W. The use of the HP improved the cooling capacity by 53% at a cold end temperature of 10 °C. An average increase of 10 W has been achieved by using HP to reduce the hot end temperature of the TE. The COP is also increased by 0.1 on average when the HP is combined into the TE. This indicates that the power consumption of the TE can be significantly reduced to provide sufficient cooling capacity for

CPU. The improvement of performance by using heat pipe is mainly due to the much lower hot end temperature and thus smaller temperature difference between the hot and cold ends. By using heat pipe to dissipate the heat from the hot end of the TE, the average reduction of hot end temperature is 10 °C.

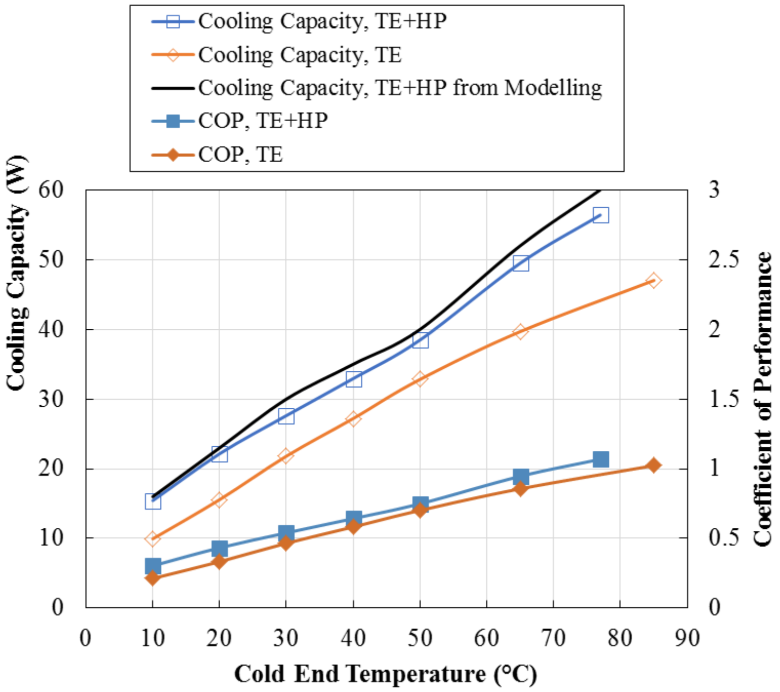

Fig. 10 Comparison of cooling capacity and COP against cold end temperature between the TE and the combined HP and TE with a voltage of 12 V

Fig. 11 compares the cooling capacity of three systems operated at 10 V: single stage TE, combination of TE and HP, and a two-stage TE. The average hot end temperature for three systems is 39 °C, 30 °C and 70 °C respectively. It can be seen that for a given cooling capacity, two stage TE can operate at much higher temperature difference between the hot and cold ends. This is similar to the conclusion made by Chen et al. **[22]**. For a temperature difference of 40 °C, there is hardly cooling for TE and combined TE and HP while there is 18 W cooling for the two-stage TE.

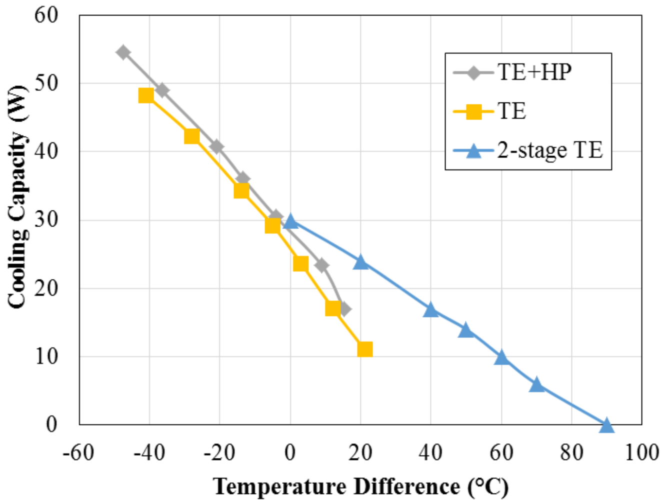

Fig. 11 Cooling capacity of the TE, combination of TE and HP, and a 2-stage TE operated at 10V

Fig. 12 plots the COP between the TE, the combined HP and TE, the two-stage TE and the VCR system against temperature difference of the cold and hot ends. The TE operates at a voltage of 10 V. It can be seen that the heat pipe doesn't change the characteristics of the thermoelectric module but only reduce the hot end temperature. The VCR system has the highest COP among these four systems. For temperature difference between 0 and 20 °C, the 2-stage TE has much higher COP than both 1-stage TE and combined TE and HP.

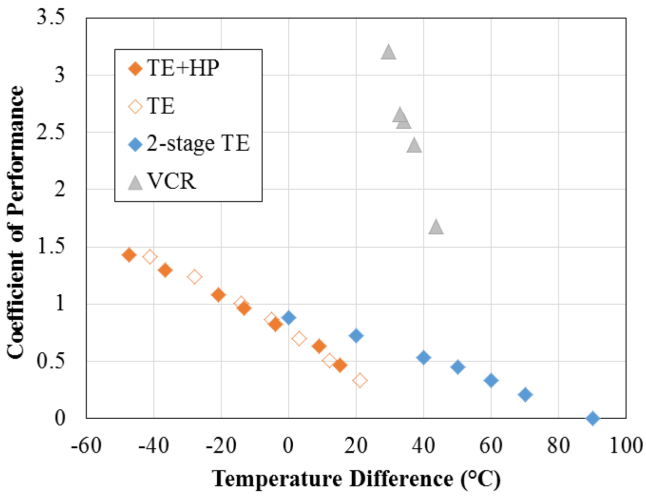

Fig. 12 Comparison of COP between the TE, the combined HP and TE, the two-stage TE and the VCR system against temperature difference between the cold and hot ends

Fig. 13 compares the cooling capacity for the HP, the TEC and the combined HP and TEC against cold end temperature. The VCR system using R134a is capable of 400 W cooling capacity at lower cold end temperature. This is 10 times the thermoelectric systems. The main concern is the size and cost. High frequency oil-free system using linear compressor and microchannel heat exchangers are very attractive for miniaturization. At a voltage of 10 V, cooling capacity was improved by 7 W on average when heat pipe assisted the thermoelectric system. Two-stage TE system shows slightly lower cooling capacity for lower temperature difference than one-stage TE but it can significantly increase the cooling at much higher temperature difference. Multi-stage TE is suitable for lower temperature applications where a moderate cooling capacity is required. Typical applications include IR detectors, CCD arrays, and electo-optics. Although heat pipe shows a much higher cooling capacity at higher cold end temperature (84 W at 51 °C), the disadvantages are the size and temperature variation. The 4 heat pipes used for assisting the thermoelectric cooler have total volume of 400 times that of the thermoelectric module itself. It is worth noting that the heat pipe for the proposed system was not an optimized design. As a passive cooling system, the surface temperature of the CPU when using heat pipe can vary significantly when heat flux varies. In contrast, the thermoelectric system can achieve steady surface temperature and much quicker cooling. More importantly much lower surface temperature that can be achieved by the thermoelectric system will further increase the reliability of the chips although typical permitted operating temperature is below 70 °C. However, it can be seen that heat pipe requires significant space and thus it can effectively be used for larger electronic devices, e.g. desktop. Thermoelectric system can be used for miniature and portable devices.

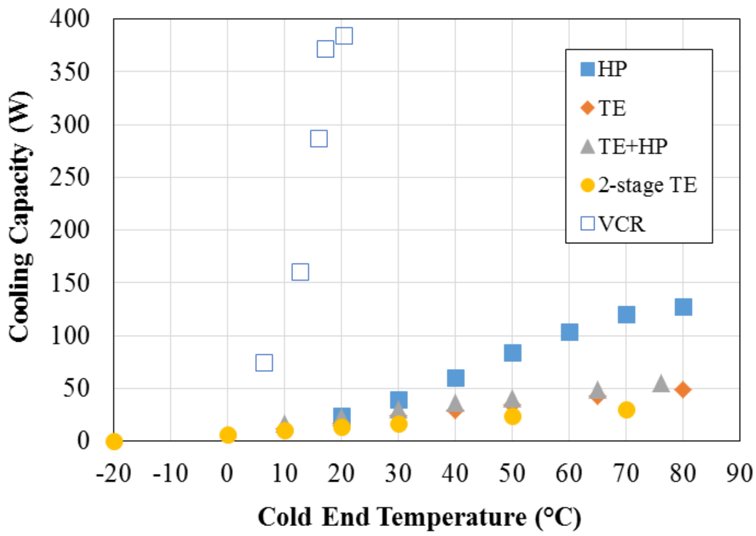

Fig. 13 Comparison of cooling capacity between HP, TE, combined HP and TE, 2-stage TE and VCR system against cold end temperature (TE operated at 10 V)

The cost of heat pipe is marginal while energy and corresponding cost savings are considerable. Thermoelectric module has higher cost than heat pipe. However, Ngondi et al. **[23]** revealed that the cost reduction of thermoelectric module coming from the raw material savings compensates for the additional manufacturing cost. Vapour compression refrigeration system consisting of oil-free linear compressor and microchannel heat exchangers for electronics cooling has the highest cost. Although microchannel heat exchangers can be made at low cost, the oil-free linear compressor is still a challenge in terms of cost reduction. Nevertheless, the vapour compression refrigeration system is much more efficient than thermoelectric system.

4.5 Experimental Uncertainty

A number of parameters were measured in the experiments, including temperature, current, voltage, electrical power and mass flow rate. Typically, a set of readings was taken every 20 minutes so as to allow time for thermal equilibrium to be attained. The measurements of temperature, current, voltage, electrical power and mass flow rate have absolute uncertainties of 2 K, 0.25%, 0.5%, 5 mW and 0.2%, respectively. The combined uncertainties of the calculated values were calculated using a 95% confidence interval. The cooling capacity and COP have absolute uncertainty values of 0.08 W and 0.02.

5. Conclusions

Five systems for electronics cooling have been evaluated numerically and experimentally in this paper. Key findings are listed as below:

(1) Heat pipe and thermoelectric models predict very close performance data to the experimental results.
(2) The thermoelectric system appears to work best at a voltage of 10 V. At a cold end temperature of 80 °C and hot end temperature of 39 °C, 10 V can provide 49 W cooling for the CPU with a COP of 1.5.
(3) The use of the HP improved the cooling capacity of the TE by 53% at a cold end temperature of 10 °C and a voltage of 10 V. The COP is also increased by 0.1 on average when HP assisted the TE. This is similar to the conclusion by Liu et al. **[15]**.
(4) The VCR system shows much higher cooling capacity and COP than any TE systems. The main concern is the cost. Heat pipe is most efficient (without power consumption) and more attractive for cooling electronics at higher temperature. However, size is main concern.
(5) Two-stage TE can significantly increase the achievable temperature difference between the hot and cold ends.

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Highlights

- Heat pipe and thermoelectric models are validated by experimental results.
- The thermoelectric system appears to work best at a voltage of 10 V.
- The heat pipe improved the cooling capacity of the thermoelectric system by 53%.
- VCR system is capable of dissipating much higher heat flux (200 W/cm2)
- Two-stage TE system can be used for cooling devices at lower temperatures.